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SPECIFICATION

SWASH PLATE TYPE VARIABLE DISPLACEMENT HYDRAULIC ROTARY MACHINE

TECHNICAL FIELD

This invention relates to a swash plate type variable displacement hydraulic rotary machine suitable for use as a hydraulic pump or motor on working vehicles such as wheel loaders, wheel type hydraulic excavator or hydraulic cranes, or crawler type hydraulic excavator or cranes.

BACKGROUND ART

Generally, swash plate type variable displacement hydraulic rotary machines have been in use on working vehicles such as wheel loaders and hydraulic excavator, as a swash plate type variable displacement hydraulic pump of a hydraulic pressure source. In other applications, swash plate type variable displacement hydraulic rotary machines have been resorted to as a hydraulic revolving motor or as a hydraulic vehicle drive motor.

According to the prior art, swash plate type variable displacement hydraulic rotary machines of this sort are

largely constituted by a tubular casing, a rotational shaft rotatably provided within the casing, a cylinder block provided in the casing for rotation with the rotational shaft and containing a plural number of axially extending cylinders in circumferentially spaced positions, a plural number of pistons reciprocally fitted in the respective cylinders of the cylinder block, a plural number of shoes attached to piston ends which are projected from the respective cylinders, a swash plate having a smooth surface on a front side to guide the respective shoes by sliding engagement therewith and tiltably supported in the casing on the back side, and a tilting actuator provided in the casing and adapted to drive the swash plate into a tilted position according to a tilting control pressure which is supplied to and from outside.

In this instance, on the back side, the swash plate is provided with a pair of convexly curved leg portions in spaced positions on the opposite sides of the rotational shaft. On the other hand, corresponding to the paired leg portions, swash plate support portions of concavely curved shape are provided on the casing thereby to tiltably support the swash plate through the paired leg portions.

Further, a pair of supply/discharge passages are provided within the casing to supply pressure oil to and from the

respective cylinders in the cylinder block. Besides, hydrostatic bearings are provided between the leg portions of the swash plate and the swash plate support portions (hereinafter referred to as "the first prior art" for brevity).

In the case of the hydrostatic bearing according to the first prior art, a part of pressure oil is led from one of the above-mentioned supply/discharge passages which is on a high pressure side to generate a dissociative force between contacting surfaces (a convexly curved surface of a leg portion of a swash plate and a tilting support surface), keeping contacting surfaces in a lubricated state by use of oil pressure (e.g., as disclosed in Japanese Patent Laid-Open No. Hei 9-166074).

Further, as a second prior art, there has been known a swash plate type variable displacement hydraulic rotary machine which is provided with first and second hydrostatic bearings independently between a pair of leg portions of a swash plate and swash plate support portions, the first hydrostatic bearing being communicated with one of supply/discharge passages which are provided in a casing, while the second hydrostatic bearing is communicated with the other one of the supply/discharge passages (e.g., as disclosed

in U. S. Patent 6,048,176).

Further, as a third prior art, there has been known a swash plate type variable displacement hydraulic rotary machine for use in a hydraulic closed circuit type hydraulic power transmission mechanism (a hydrostatic transmission, hereinafter referred to as "HST" for brevity). This swash plate type variable displacement hydraulic rotary machine is provided with a swash plate and a tilting actuator which drives the swash plate. The tilting actuator is adapted to tilt the swash plate in a forward or reverse direction from a zero angle neutral position, for example, by switching a pressure oil delivery direction of a hydraulic pump from forward to reverse direction or vice versa (e.g., as disclosed in Japanese Pstent Laid-Open No. Sho 63-259182).

By the way, in the case of the first prior art mentioned above, the hydrostatic bearings which are provided between leg portions of a swash plate and swash plate support portions of a casing is supplied with pressure oil from one of the supply/discharge passages, so that, due to pressure variations in the supply/discharge passages, unbalanced situations take place between the hydraulic reaction forces imposed on the swash plate by the respective pistons (pressures imposed on the swash plate by piston reaction forces) and dissociative

forces of hydrostatic bearings.

Under unbalanced situations, each leg portion of the swash plate tends to float up in a tilted state away from the swash plate support portion, letting pressure oil in the hydrostatic bearing leak to outside. As a result, the hydrostatic bearing becomes unable to maintain a lubricated state between each leg portion of the swash plate and a swash plate support portion.

Further, in the case of a hydraulic motor, for example, its rotational shaft is put in rotation in a forward or reverse directions, and every time when the rotational direction of the rotational shaft is changed, a pair of supply/discharge passages are switched from a higher pressure side to a lower pressure side or from a lower pressure side to a higher pressure side. That is to say, the above-mentioned first prior art has a serious problem that the pressure in the hydrostatic bearing is varied abruptly every time the rotational direction of the rotational shaft is changed, and, under such circumstances, the hydrostatic bearing does not function as a hydrostatic bearing any longer.

In the case of the swash plate type variable displacement hydraulic pump of the above-mentioned third prior art, for use with an HST or the like, a swash plate is tilted in forward

and reverse directions from a zero angle neutral position by means of a tilting actuator. The swash plate type variable displacement hydraulic pump of the third prior art suffers from the same problem even if the hydrostatic bearings of the first prior art are applied by switching a pair of supply/discharge passages from a higher pressure side to a lower pressure side or from a lower pressure side to higher pressure side corresponding to the tilted direction of the swash plate.

On the other hand, in the case of the hydraulic rotary machine of the second prior art mentioned above, first and second hydrostatic bearings are independently provided between a pair of leg portions of a swash plate and a pair of tilting support surfaces which are formed on swash plate support portions, communicating the first hydrostatic bearing with one of paired supply/discharge passages while communicating the second hydrostatic bearing with the other one of supply/discharge passages.

Therefore, the hydraulic rotary machine of the second prior art can be applied as a hydraulic motor with a reversible rotational shaft or as a swash plate type variable displacement hydraulic pump for use in HST.

However, in the case of the hydraulic rotary machine of

the second prior art, difficulties are encountered in balancing dissociative forces of the first and second hydrostatic bearings with pressures imposed on the swash plate by piston reaction forces at different radial positions on the opposite sides (both sides of radial positions) of the rotational shaft. Therefore, the leg portions of the swash plate can be inclined and floated up away from swash plate support portions.

As explained above, even the hydraulic rotary machine of the second prior art has the problem that pressure oil which is led in the first and second hydrostatic bearings is liable to leak to outside. In the event of pressure oil leakage, it becomes difficult to maintain a lubricated state between the leg portions of the swash plate and the swash plate support portions.

DISCLOSURE OF THE INVENTION

In view of the above-discussed problems with the prior art, it is an object of the present invention to provide a swash plate type variable displacement hydraulic rotary machine which is particularly arranged to guarantee stabilized performance of hydrostatic bearings by adjusting the balance between dissociative forces of hydrostatic bearings and

pressures which are exerted on a swash plate by piston reaction forces.

It is another object of the present invention to provide a swash plate type variable displacement hydraulic rotary machine of high versatility, which can be applied as a hydraulic motor with a reversible rotational shaft or as a swash plate type variable displacement hydraulic pump for use, for example, in an HTS, and which can be manufactured with a higher productivity and at a reduced cost.

(1) According to the present invention, in order to achieve the above-stated objectives, there is provided a swash plate type variable displacement hydraulic rotary machine comprising a tubular casing having at one end a swash plate support portion and at the other end a pair of supply/discharge passages, a rotational shaft rotatably supported in the casing, a cylinder block provided in the casing for rotation together with the rotational shaft and containing a plural number of axially extending cylinders in circumferentially spaced positions, a plural number of pistons reciprocally fitted in the cylinders of the cylinder block, a plural number of shoes attached to projected ends of the pistons projected from the respective cylinders, a swash plate having on a front side a smooth surface for guiding the shoes

by sliding contact therewith and on a rear side a pair of leg portions tiltably supported by the swash plate support portion, a tilting actuator provided within the casing to drive the swash plate into a tilted position according to a tilting control pressure which is supplied to and from outside, hydrostatic bearings provided between the leg portions of the swash plate and the swash plate support portion in communication with the supply/discharge passages to keep contacting surfaces of the leg portions and the swash plate support portion in a lubricated state.

The swash plate type variable displacement hydraulic rotary machine according to the present invention is characterized in that the hydrostatic bearings comprise: the hydrostatic bearing constituted by a first main hydrostatic bearing provided on one leg portion of the pair of leg portions, a second main hydrostatic bearing provided on the other leg portion, a first auxiliary hydrostatic bearing provided on the other leg portion at a position spaced from the second main hydrostatic bearing, and a second auxiliary hydrostatic bearing provided on the one leg portion at a position spaced from the first main hydrostatic bearing.

With the above arrangements according to the present invention, no matter which one of supply/discharge passages is

at a higher pressure, dissociative forces are generated between leg portions of the swash plate and the swash plate support member by the main and auxiliary hydrostatic bearings in a well balanced state relative to hydraulic reaction forces (pressures of piston reaction forces) which are imposed on the swash plate by pistons, guaranteeing stabilized performance as hydrostatic bearings.

Thus, the swash plate type variable displacement hydraulic rotary machine according to the invention can be easily applied to hydraulic rotary machines in which a pair of supply/discharge passages are reversibly switched to a high pressure or to a low pressure (e.g., a hydraulic motor having a reversible rotational shaft or a swash plate type variable displacement hydraulic pump for use in HST). It follows that the swash plate type variable displacement hydraulic rotary machine of the present invention is enhanced in versatility and can be manufactured with a higher productivity and at a reduced cost.

(2) Further, according to the present invention, the first main hydrostatic bearing is located in the vicinity of an acting point of a resultant force of hydraulic reaction forces exerted on the swash plate by the pistons radially on one side of the rotational shaft, and the second main

hydrostatic bearing is located in the vicinity of an acting point of a resultant force of hydraulic reaction forces exerted on the swash plate by the pistons radially on the other side of the rotational shaft.

By locating in this manner the first and second main hydrostatic bearings at a position in the vicinity of an acting point of a resultant force of hydraulic reaction forces which are exerted on the swash plate by the pistons, it becomes possible to locate acting points of dissociative forces of the main hydrostatic bearings near acting points of resultant forces of hydraulic reaction forces (piston reaction forces) which are exerted on the swash plate by the pistons on the side of the cylinder block. This arrangement makes it possible to suppress a moment (e.g., a moment about a acting point of a resultant force of hydraulic reaction forces) which acts on the swash plate as a result of actions of hydraulic reaction forces and dissociative forces of the hydrostatic bearings. As a result, it becomes possible to minimize effective bearing surface areas of the first and second auxiliary hydrostatic bearing and downsize the hydraulic rotary machine as a whole including the swash plate.

(3) Further, according to the present invention, the swash plate is provided with a through hole between the pair

of leg portions to insert the rotational shaft leaving a gap therearound, and the first and second main hydrostatic bearings are located closer to the through hole and have a larger effective bearing surface area as compared with the first and second auxiliary hydrostatic bearings.

Even with the arrangements just described, acting points of dissociative forces on the swash plate by the main hydrostatic bearings can be located in the proximity of acting points of a resultant force of hydraulic reaction forces which are exerted on the swash plate by the respective pistons. As a consequence, it becomes possible to suppress a moment which is exerted on the swash plate by the hydraulic reaction forces and dissociative forces, to minimize effective bearing surface areas of the first and second auxiliary hydrostatic bearings, and to downsize the hydraulic rotary machine as a whole including the swash plate.

(4) Further, according to the present invention, first and second slide bearings are provided on the pair of leg portions at positions more radially distant from the rotational shaft than the first and second main hydrostatic bearings and the first and second auxiliary hydrostatic bearings.

In this case, even if changes occur to the balance of

moments which act on the swash plate due to pressure variations on the side of the supply/discharge passages, stability of the swash plate can be secured by the first and second slide bearings. Beside, the provision of the first and second slide bearings contributes to lessen surface pressures between the leg portions of the swash plate and the tilting support surfaces of the swash plate support member, suppressing abrasive wear of contacting surfaces to guarantee higher operational reliability and a prolonged service life.

(5) On the other hand, according to the present invention, the first main hydrostatic bearing and the first auxiliary hydrostatic bearing are communicated with one of the supply/discharge passages through an oil passage, while the second main hydrostatic bearing and the second auxiliary hydrostatic bearing are communicated with the other one of the supply/discharge passages through another oil passage.

Consequently, when one of the supply/discharge passages turns to a higher pressure than the other supply/discharge passage, high pressure oil is supplied to the first main hydrostatic bearing on one leg portion of the swash plate and at the same time, high pressure oil is supplied to the first auxiliary hydrostatic bearing on the other leg portion of the swash plate. When the other one of the supply/discharge

passages turns to a higher pressure than one supply/discharge passage, high pressure oil is supplied to the second auxiliary hydrostatic bearing on one leg portion of the swash plate and at the same time, high pressure oil is supplied to the second main hydrostatic bearing on the other leg portion. Thus, even when one of a pair of supply/discharge passages turns to a higher pressure, dissociative forces are generated between the leg portions of the swash plate and the swash plate support member by the main and auxiliary hydrostatic bearings in a well balanced state against hydraulic reaction forces which are exerted on the swash plate by the pistons, ensuring stabilized performance as hydrostatic bearings.

(6) Further, according to the present invention, the first main hydrostatic bearing and the first auxiliary hydrostatic bearing are communicated with one of the supply/discharge passages through an oil passage having a throttle for commonly adjusting an amount of pressure oil to be supplied to the first main and auxiliary hydrostatic bearings, while the second main hydrostatic bearing and the second auxiliary hydrostatic bearing are communicated with the other one of the supply/discharge passages through another oil passage having an another throttle for commonly adjusting an amount of pressure oil to be supplied to the second main and

auxiliary hydrostatic bearings.

In this manner, by means of the common throttle which is provided in the course of an oil passage which communicates the first main and auxiliary hydrostatic bearings with one of the supply/discharge passages, an amount of pressure oil to be supplied to the first main and auxiliary hydrostatic bearings is commonly adjusted, thereby permitting to increase or decrease dissociative forces of the swash plate by these hydrostatic bearings according to the amount of pressure oil supply. Similarly, by means of providing another common throttle in the course of another oil passage which communicates the second main and auxiliary hydrostatic bearings with the other supply/discharge passage, an amount of pressure oil to be supplied to the second main and auxiliary hydrostatic bearings is commonly adjusted, thereby permitting to increase or decrease dissociative forces of the swash plate by these hydrostatic bearings according to the amount of pressure oil supply.

(7) Further, according to the present invention, the first main hydrostatic bearing and the first auxiliary hydrostatic bearing are communicated with one of the supply/discharge passages through oil passages each having discrete throttles for adjusting an amount of pressure oil to

be supplied to the first main and auxiliary hydrostatic bearings separately and independently of each other, while the second main hydrostatic bearing and the second auxiliary hydrostatic bearing are communicated with the other one of the supply/discharge passages through the other oil passages each having another discrete throttles for adjusting an amount of pressure oil to be supplied to the second main and auxiliary hydrostatic bearings separately and independently of each other.

In this manner, by providing discrete throttles in the course of oil passages which communicate the first main hydrostatic bearing and the first auxiliary hydrostatic bearing with one supply/discharge passage, an amount of pressure oil to be supplied to the first main and auxiliary hydrostatic bearings can be adjusted separately and independently of each other, thereby permitting to increase or decrease dissociative forces of the swash plate by these main and auxiliary hydrostatic bearings according to each amount of pressure oil supply. Similarly, by providing another discrete throttles in the course of the other oil passages which communicate the second main hydrostatic bearing and the second auxiliary hydrostatic bearing with the other supply/discharge passage, an amount of pressure oil to be supplied to the

second main and auxiliary hydrostatic bearings can be adjusted separately and independently of each other, thereby permitting to increase or decrease dissociative forces of the swash plate by these main and auxiliary hydrostatic bearings according to an amount of pressure oil supply. That is to say, moments which act on the swash plate are well balanced by dissociative forces and hydraulic reaction forces which are exerted by pistons, for stabilizing the swash plate and ensuring higher reliability and prolonged service life as a swash plate type hydraulic rotary machine.

(8) Further, according to the present invention, the first main hydrostatic bearing and the first auxiliary hydrostatic bearing are communicated with the one supply/discharge passage by way of a common oil passage being in communication with the one supply/discharge passage at one end and extended toward the first main and auxiliary hydrostatic bearings at the other end, and branched oil passages provided at the other end of the common oil passage and connected separately to the first main hydrostatic bearing and the first auxiliary hydrostatic bearing; and the second main hydrostatic bearing and the second auxiliary hydrostatic bearing are communicated with the other supply/discharge passage by way of another common oil passage being in

communication with the other supply/discharge passage at one end and extended toward the second main and auxiliary hydrostatic bearings at the other end, and another branched oil passages provided at the other end of the common oil passages and connected separately to the second main hydrostatic bearing and the second auxiliary hydrostatic bearing.

In this case, a common oil passage and branched oil passages are provided between one supply/discharge passage and the first main and auxiliary hydrostatic bearings, and similarly another common passage and another branched oil passages are provided between the other supply/discharge passage and the second main and auxiliary hydrostatic bearings. Therefore, for example, as compared with a case where discrete oil passages are provided for the respective hydrostatic bearings, it becomes possible to reduce the number of oil passages in a casing to realize a machine of compact and simple construction which can be manufactured with higher productivity and at a lower cost.

(9) Further, according to the present invention, a common throttle is provided in the common oil passage thereby to adjust an amount of pressure oil to be supplied from the one supply/discharge passage to the first main and auxiliary

hydrostatic bearings, and discrete throttles are provided in the branched oil passages thereby to separately and independently adjust amounts of pressure oil to be supplied to the first main hydrostatic bearing and the first auxiliary hydrostatic bearing; and the other common throttle is provided in the another common oil passage thereby to adjust an amount of pressure oil to be supplied from the other supply/discharge passage to the second main and auxiliary oil passages, and another discrete throttles are provided in the branched oil passages thereby to separately and independently adjust amounts of pressure oil to be supplied to the second main hydrostatic bearing and the second auxiliary hydrostatic bearing.

In this manner, a common throttle is provided in the course of common oil passage which is located upstream of the branched oil passages, and a discrete throttle is provided in the course of each one of the branched oil passages. Therefore, even in a case where the common throttle is of a relatively large throttle diameter (orifice diameter), an amount of pressure oil to be supplied to the main and auxiliary hydrostatic bearings can be adjusted suitably by way of the common throttle, lessening possibilities of the common throttle being blocked (clogged) with foreign matter like

dusts and thus improving operational reliability of the machine. Further, even if a fine gap space exists around each hydrostatic bearing, the common throttle has an effect of suppressing pressure oil leaks through the fine gap space, contributing to facilitate a machining process in addition to higher productivity and a reduced production cost.

(10) Further, according to the present invention, the swash plate is driven by the tilting actuator to tilt in both forward and reverse directions from zero angle neutral position. As a consequence, even in case the hydraulic rotary machine of the invention is applied as a swash plate type variable displacement hydraulic pump for HST, and connecting the hydraulic pump to a hydraulic actuator through a closed hydraulic circuit, the discharged direction of pressure oil can be reversibly switched and controlled according to the tilting direction of the swash plate (in a forward or reverse direction). No matter the swash plate is tilted in a forward or reverse direction, the swash plate can be put in a stabilized tilting motion and maintain favorably lubricated state between the leg portion of the swash plate and the swash plate support member.

(11) Further, according to the present invention, the swash plate type variable displacement hydraulic rotary

machine further comprises within the casing a regulator in the form of a servo valve having a spool within a control sleeve and adapted to supply the tilting control pressure to and from the tilting actuator in response to an external command signal, and a feedback mechanism adapted to feed back the control sleeve of the regulator according to a tilting movements of the swash plate; the feedback mechanism comprising: a conversion mechanism adapted to convert the tilting movements of the swash plate into an axial displacement as being located in an initial position at one axial end along the rotational shaft when the swash place is in a neutral position, and as being displaced toward the other axial end from the initial position when the swash plate is driven to tilt in forward or reverse direction, and a displacement transmission member located between the conversion mechanism and the control sleeve of the regulator to transmit the axial displacement converted by the conversion mechanism to the control sleeve of the regulator.

With the arrangements just described, when the swash plate is driven by the tilting actuator to tilt in a forward or reverse direction, the control sleeve of the regulator is put in a sliding displacement in the same direction as the spool by feedback control of the regulator. No matter the

swash plate is tilted in a forward or reverse direction, the regulator can be put in operation smoothly under feedback control. Besides, since the regulator can be constituted by a servo valve having a spool within a control sleeve, it becomes possible to provide a swash plate type variable displacement hydraulic rotary machine which is as a whole simplified in construction.

BRIEF DESCRIPTION OF THE DRAWINGS

In the accompanying drawings:

Fig. 1 is a diagram of a vehicle drive hydraulic circuit of a wheel type working vehicle, providing a swash plate type variable displacement hydraulic pump according to a first embodiment of the present invention;

Fig. 2 is a longitudinal sectional view of the hydraulic pump in Fig. 1;

Fig. 3 is a longitudinal sectional view of the hydraulic pump taken from the direction of arrows III-III in Fig. 2;

Fig. 4 is an enlarged sectional view of the hydraulic pump in Fig. 3;

Fig. 5 is a sectional view showing a swash plate support member and a swash plate in Fig. 4 on an enlarged scale along with a hydrostatic bearing;

Fig. 6 is an enlarged sectional view of the swash plate in a neutral position, taken from the direction of arrows VI-VI in Fig. 4;

Fig. 7 is a sectional view of the swash plate which has been tilted in a forward direction, taken in the same position as Fig. 6;

Fig. 8 is a enlarged perspective view of the swash plate in Fig. 3;

Fig. 9 is a back view of the swash plate taken from the rear side in Fig. 8;

Fig. 10 is a circuit diagram of a tilting controller of the swash plate of the first embodiment;

Fig. 11 is a front view showing the swash plate in Fig. 10 along with tilting pistons;

Fig. 12 is a front view of the swash plate of Fig. 11, which has been tilted in a forward direction;

Fig. 13 is a front view of the swash plate of Fig. 11, which has been tilted in a reverse direction;

Fig. 14 is a longitudinal sectional view of a hydraulic pump according to a second embodiment of the present invention, taken in the same position as Fig. 3;

Fig. 15 is an enlarged perspective view of a swash plate in Fig. 14; and

Fig. 16 is a back view of the swash plate taken from rear side in Fig. 15.

BEST MODE FOR CARRYING OUT THE INVENTION

Hereafter, with reference to the accompanying drawings, the swash plate type variable displacement hydraulic rotary machine of the present invention is described more particularly by way of its preferred embodiments which are applied by way of example to a vehicle drive hydraulic circuit of a wheel type working vehicle such as a wheel loader or the like.

Shown in Figs. 1 through 13 is a swash plate type variable displacement hydraulic rotary machine according to a first embodiment of the present invention.

In the drawings, indicated at 1 is a swash plate type hydraulic pump as the swash plate type variable displacement hydraulic rotary machine according to the invention. The swash plate type hydraulic pump 1 is constituted by a casing 11, a rotational shaft 13, a cylinder block 14, a plural number of cylinders 15, pistons 16, shoes 17, a valve plate 19, a swash plate support member 20, and a swash plate 21, which will be described hereinafter.

Further, the rotational shaft 13 of the hydraulic pump 1

is rotationally driven by a prime mover 2, for example, by a diesel engine or the like serving as a drive source, to deliver pressure oil to a pair of main conduits 3A and 3B as shown in Fig. 1. Through the main conduits 3A and 3B, the hydraulic pump 1 is connected to a hydraulic motor 5, which will be described hereinafter, to form the so-called closed hydraulic circuit 4.

Indicated at 5 is a vehicle drive hydraulic motor serving as a hydraulic actuator. This hydraulic motor 5 is coupled, for example, with wheels 7 of a wheel type working vehicle through a reducer 6. As pressure oil is supplied to and from the hydraulic pump 1 through the main conduits 3A and 3B, the wheels 7 of the working vehicle are rotationally driven by the hydraulic motor 5 to put the working vehicle in travel.

Indicated at 11 is a tubular casing which forms an outer shell of the hydraulic pump 1. As shown in Figs. 2 to 4, the casing 11 is composed of a tubular main casing 11A, and front and rear casings 11B and 11C which close the opposite ends of the main casing 11A.

Further, as shown in Fig. 3, a slot 11D and a drain passage 11E are formed in an outer peripheral side of the main casing 11A. Through the slot 11D and drain passage 11E, the inner cavity of the main casing 11A is constantly communicated

with a valve housing 35 of a regulator 34 which will be described in greater detail hereinafter. A translation bar 44 is slidably received in the slot 11D of the main casing 11A through a guide member 45 as described hereinafter. The inner cavity of the main casing 11 forms the so-called drain chamber which is connected to a tank 47, which will be described later on.

In this instance, as shown in Figs. 2 to 4, a swash plate support member 20 is located on the inner side of the front casing 11B at one end of the main casing 11A, in face to face relation with a rear side of the swash plate 21 as described hereinafter. Further, a pair of supply/discharge passages 12A and 12B are provided in the rear casing 11C at the other end of the main casing 11A, and connected to the main conduits 3A and 3B of Fig. 1, respectively.

Indicated at 13 is a rotational shaft which is rotatably provided within the casing 11, more specifically, rotatably supported on the front and rear casings 11B and 11C through bearings, respectively. One end of the rotational shaft 13 is axially projected out of the front casing 11B, and an axially projected end 13A of the rotational shaft 13 is rotationally driven by the prime mover 2 of Fig. 1.

Denoted at 14 is a cylinder block which is provided

within the casing 11 and put in rotation together with the rotational shaft 13. Provided in the cylinder block 14 is a plural number of cylinders 15 which are located at intervals in the circumferential direction and extended to axis direction.

Indicated at 16 are pistons which are slidably fitted in the cylinders 15 of the cylinder block 14. As the swash plate 21 is tilted in a forward or reverse direction, these pistons 16 are reciprocated within the respective cylinders 15 in step with rotation of the cylinder block 14, repeating a cycle of intake and discharge stages.

Indicated at 17 are shoes which are provided on the respective pistons 16. These shoes 17 are swingably attached to axial ends (projected ends) of the pistons 16 which are projected out of the respective cylinders 15 of the cylinder block 14 to axial direction of the rotational shaft 13.

Designated at 18 is an annular shoe holder which serves to hold the respective shoes 17 against the swash plate 21. As shown in Figs. 3 to 7, this shoe holder 18 presses the respective shoes 17 against a smooth surface 21C of the swash plate 21, which will be described hereinafter, and putting each shoe 17 in sliding displacement on and along the smooth surface 21C of the swash plate 21 in such a way as to draw an

annular locus of movement.

Indicated at 19 is a valve plate which is provided within the casing 11, at a position between the rear casing 11C and the cylinder block 14. This valve plate 19 is held in sliding contact with an end face of the cylinder block 14 to support the cylinder block 14 for rotation together with the rotational shaft 13. Further, as shown in Figs. 3 and 4, a pair of eyebrow-shape supply/discharge ports 19A and 19B are formed in the valve plate 19. These supply/discharge ports 19A and 19B are constantly communicated with the supply/discharge passages 12A and 12B of the rear casing 11C.

In this instance, as the cylinder block 14 is put in rotation, these supply/discharge ports 19A and 19B of the valve plate 19 are intermittently communicated with the respective cylinders 15. These supply/discharge ports 19A and 19B have functions of taking operating oil into the respective cylinders 15 from one of the supply/discharge passages 12A or 12B for pressurization by the pistons 16, while discharging high pressurized operating oil in the respective cylinders 15 to the other one of the supply/discharge passages 12B or 12A.

Indicated at 20 is a swash plate support member which functions as a support portion for the swash plate 21. This swash plate support member 20 is provided on the front casing

11B in such a way as to be located to the circumference of the rotational shaft 13. As shown in Fig. 4, this swash plate support member 20 is provided with a pair of tilting support surfaces 20A and 20B, for example, on the right and left sides of the rotational shaft 13 thereby to tiltably support the swash plate 21.

The tilting support surfaces 20A and 20B of the swash plate support member 20 are formed in a concavely curved shape correspondingly to leg portions 21A and 21B of the swash plate 21 to guide the swash plate 21 tiltably in the directions of arrows A and B about a tilting center C as exemplified in Figs. 6 and 7. Further, bored in the swash plate support member 20 are part of branched oil passages 24B, 24C, 25B and 25C, which will be described hereinafter.

Indicated at 21 is the swash plate which is adopted in the present embodiment of the invention. This swash plate 21 is tiltably provided within the casing 11 through the swash plate support member 20. As shown in Figs. 2 to 7, on the rear side of the swash plate 21, a pair of right and left leg portions 21A and 21B are projectionally provided in a convexly curved shape toward the tilting support surfaces 20A and 20B of the swash plate support member 20. The leg portions 21A and 21B of the swash plate 21 are provided in radially spaced

positions, for example, on the opposite sides of the rotational shaft 13, and slidably fitted in the concavely curved tilting support surface 20A and 20B on the swash plate support member 20.

On the other hand, as shown in Figs. 2 to 7, a smooth surface 21C is provided on the front side of the swash plate 21 thereby to guide sliding movements of the shoes 17. Further, an axial through hole 21D is bored through the swash plate 21. The rotational shaft 13 is received with gapped space in the through hole 21D between the right and left leg portions 21A and 21B of the swash plate 21.

In this instance, as shown in Figs. 6 to 10, the leg portions 21A and 21B of the swash plate 21 are formed in an arc of a radius R from the tilting center C which is located on the axis O-O of the rotational shaft 13. As shown in Figs. 6 and 11, the swash plate 21 is driven to tilt in a forward direction (in the direction of arrow A) or in a reverse direction (in the direction of arrow B) from a zero angle neutral position by means of tilting actuators 32 and 33, which will be described hereinafter. At this time, the capacity (the volume of pressure oil delivery) of the hydraulic pump 1 is variably controlled by the tilting angle θ of the swash plate 21.

The swash plate 21 receives hydraulic reaction forces (piston reaction forces) from the pistons 16 which are put in rotation together with the cylinder block 14 around the rotational shaft 13. In this case, as the cylinder block 14 is put in rotation, acting points (hereinafter referred to as "resultant force acting points k1 and k2") of resultant forces f1 and f2 of the hydraulic reaction forces change positions in the fashion of a figure "∞" as shown in Fig. 9. When tilted in a forward direction from a neutral position, the swash plate 21 receives hydraulic reaction forces at the position of the resultant force acting point k1, and, when tilted in a reverse direction from a neutral position, the swash plate 21 receives hydraulic reaction forces at the position of the resultant force acting point k2.

Indicated at 22 are hydrostatic bearings which are provided between the tilting support surfaces 20A and 20B of the swash plate support member 20 and the leg portions 21A and 21B of the swash plate 21, respectively. By means of pressure oil which is led from the supply/discharge passages 12A and 12B in the rear casing 11C, the hydrostatic bearings 22 generate a dissociative force (a hydraulic pressure force) between the tilting support surfaces 20A and 20B and the leg portions 21A and 21B of the swash plate 21 to maintain

contacting surfaces in a lubricated state.

As shown in Figs. 5, 8 and 9, the hydrostatic bearings 22 are composed of a first main hydrostatic bearing 22A which is provided on a convexly curved surface of one leg portion 21A at a position in the proximity of the through hole 21D of the swash plate 21, a second main hydrostatic bearing 22B which is provided on a convexly curved surface of the other leg portion 21B at a position in the proximity of the through hole 21D of the swash plate 21, a first auxiliary hydrostatic bearing 22C which is provided on the convexly curved surface of the leg portion 21B at a radially spaced position from the second main hydrostatic bearing 22B, and a second auxiliary hydrostatic bearing 22D which is provided on the convexly curved surface of the leg portion 21A at a radially spaced position from the first main hydrostatic bearing 22A.

Of the hydrostatic bearings 22A to 22D, the first main hydrostatic bearing 22A and the first auxiliary hydrostatic bearing 22C are connected to one supply/discharge passage 12A through an oil guide passage 24 which will be described hereinafter. The second main hydrostatic bearing 22B and the second auxiliary hydrostatic bearing 22D are connected to the other supply/discharge passage 12B through an oil guide passage 25 which will also be described later on.

In this instance, as shown in Fig. 8, the first and second main hydrostatic bearings 22A and 22B are formed as grooves extending in the directions of arrows A and B along convex curved surfaces of the leg portions 21A and 21B, respectively, each presenting a narrow rectangular shape in plane view as seen in Fig. 9. Further, in reference to the through hole 21D of the swash plate 21, the first and second auxiliary hydrostatic bearings 22C and 22D are located in positions radially on the outer sides of the first and second main hydrostatic bearings 22A and 22B, respectively.

The first and second auxiliary hydrostatic bearings 22C and 22D are also formed as grooves extending along concavely curved surfaces of the leg portions 21A and 21B substantially in parallel relations with the first and second main hydrostatic bearings 22A and 22B (in the directions of arrows A and B in Fig.8), each presenting a narrow rectangular shape in plane view as seen in Fig. 9. However, the first and second auxiliary hydrostatic bearings 22C and 22D are smaller in length (groove length in the direction of arrows A and B) and width as compared with the first and second main hydrostatic bearings 22A and 22B.

More specifically, the first main hydrostatic bearing 22A is located in a position in the proximity of the resultant

force acting point k1 of hydraulic reaction forces which are exerted on the swash plate 21 by pistons 16 radially on one side of the through hole 21D (on the right side in Fig. 9) and at a distance La from the resultant force acting point k1. Further, the second main hydrostatic bearing 22B is located in a position in the proximity of the resultant force acting point k2 of hydraulic reaction forces which are exerted on the swash plate 21 by pistons 16 radially on the other side of the through hole 21D (on the left side in Fig. 9) and at a distance Lb from the resultant force acting point k2.

Further, the first auxiliary hydrostatic bearing 22C is located in a position at a distance Lc (Lc > La) from the resultant force acting point k1 of hydraulic reaction forces which are exerted on the swash plate 21 by pistons 16 radially on the above-mentioned the other side of the through hole 21D (on the left side in Fig. 9). The second auxiliary hydrostatic bearing 22D is located in a position at a distance Ld (Ld > Lb) from the resultant force acting point k2 of hydraulic reaction forces which are exerted on the swash plate 21 by pistons 16 radially on one side of the through hole 21D (on the right side in Fig. 9).

As shown in Figs. 5 and 9, the first and second main hydrostatic bearings 22A and 22B are located closer to the

through hole 21D than the first and second auxiliary hydrostatic bearings 22C and 22D. Further, as expressed by Formulas (4) and (8) which will be given hereinafter, the first and second main hydrostatic bearings 22A and 22B have effective bearing surface areas Sa and Sb which are broader than effective bearing surface areas Sc and Sd of the first and second auxiliary hydrostatic bearings 22C and 22D. The effective bearing surface areas Sa, Sb, Sc and Sd are equivalent to pressure receiving surface areas of the bearings 22A, 22B, 22C and 22D, respectively.

Indicated at 23A and 23B are first and second slide bearings which are provided on the leg portions 21A and 21B of the swash plate 21. As shown in Figs. 5, 8 and 9, these first and second slide bearings 23A and 23B are located on the opposite sides of the through hole 21D and in positions more distant from the through hole 21D than the main hydrostatic bearings 22A and 22B and the auxiliary hydrostatic bearings 22C and 22D in the radial direction. Namely, as shown in Fig. 8, the slide bearings 23A and 23B are formed in a convexly curved shape at right and left outer edges of the leg portions 21A and 21B, respectively.

The slide bearings 23A and 23B are slidably held in contact with the tilting support surfaces 20A and 20B of the

swash plate support member 20 under a small surface pressure. Thus, in cooperation with the hydrostatic bearings 22A to 22D, the slide bearings 23A and 23B guarantee smooth tilting movements of the leg portions 21A and 21B of the swash plate 21 along the swash plate support member 20.

Indicated at 24 is an oil guide passage for leading pressure oil to the first main hydrostatic bearing 22A and the first auxiliary hydrostatic bearing 22C of the hydrostatic bearing 22. As shown in Figs. 4 and 5, the oil guide passage 24 is provided between the supply/discharge passage 12A and the first main and auxiliary hydrostatic bearings 22A and 22C. In this instance, the oil guide passage 24 is provided in the casing 11, and composed of a common oil passage 24A which is connected at one end with the supply/discharge passage 12A and extended at the other end toward the first main and auxiliary hydrostatic bearings 22A and 22C, and two branched oil passages 24B and 24C which are branched from the other end of the common oil passage 24A. One branched oil passage 24B is connected to the first main hydrostatic bearing 22A, while the other branched oil passage 24C is connected to the first auxiliary hydrostatic bearing 22C.

The each branched oil passages 24B and 24C of the oil guide passage 24 are branched and extended into the swash

plate support member 20 from the side of the front casing 11B of the casing 11. The extended fore end of the branched oil passage 24B is opened into the first main hydrostatic bearing 22A on the side of the tilting support surface 20A of the swash plate support member 20. The extended fore end of the branched oil passage 24C is opened into the first auxiliary hydrostatic bearing 22C on the side of the tilting support surface 20B of the swash plate support member 20.

Indicated at 25 is an another oil guide passage for leading pressure oil to the second main hydrostatic bearing 22B and the second auxiliary hydrostatic bearing 22D of the hydraulic bearing 22. As shown in Figs. 4 and 5, this oil guide passage 25 is provided between the other supply/discharge passage 12B and the second main and auxiliary hydrostatic bearings 22B and 22D. In this instance, the oil guide passage 25 is provided in the casing 11, and composed of a common oil passage 25A which is connected at one end with the supply/discharge passage 12B and extended at the other end toward the second main and auxiliary hydrostatic bearings 22B and 22D, and two branched oil passages 25B and 25C which are branched from the other end of the common oil passage 25A respectively. One branched oil passage 25B is connected to the second main hydrostatic bearing 22B, while the other

branched oil passage 25C is connected to the second auxiliary hydrostatic bearing 22D.

The each branched oil passages 25B and 25C of the oil guide passage 25 are branched and extended into the swash plate support member 20 from the side of the front casing 11B of the casing 11. The extended fore end of the branched oil passage 25B is opened into the second main hydrostatic bearing 22B on the side of the tilting support surface 20B of the swash plate support member 20. The extended fore end of the branched oil passage 25C is opened into the second auxiliary hydrostatic bearing 22D on the side of the tilting support surface 20A of the swash plate support member 20.

Indicated at 26 is a common throttle which is provided in the course of the common oil passage 24A, and at 27 is another common throttle which is provided in the course of the common oil passage 25A. Of the two common throttles 26 and 27 which are shown in Figs. 4 and 5, one common throttle 26 functions to adjust the amount of pressure oil to be commonly supplied to the first main and auxiliary hydrostatic bearings 22A and 22C from the supply/discharge passage 12A, according to its throttling diameter (orifice diameter). The other common throttle 27 functions to adjust the amount of pressure oil to be commonly supplied to the second main and auxiliary

hydrostatic bearings 22B and 22D from the supply/discharge passage 12B, according to its throttling diameter (orifice diameter).

More particularly, the common throttles 26 and 27 have a larger throttling diameter than discrete throttles 28 to 31 which will be described hereinafter, and function to roughly adjust the amount of pressure oil to be supplied to the main hydrostatic bearings 22A and 22B and to the auxiliary hydrostatic bearings 22C and 22D from the supply/discharge passages 12A and 12B. Thus, the common throttles 26 and 27 function to suppress large variations in the amount of pressure oil to be supplied to the main hydrostatic bearings 22A and 22B and the auxiliary hydrostatic bearings 22C and 22D.

Indicated at 28 and 29 are throttles (hereinafter referred to simply as discrete throttles 28 and 29) which are provided in the course of the branched oil passages 24B and 25B, respectively, and indicated at 30 and 31 are other throttles (hereinafter referred to simply as discrete throttles 30 and 31) which are provided in the course of the branched oil passages 24C and 25C, respectively. In this instance, the discrete throttles 28 to 31 have a smaller throttling diameter than the common throttles 26 and 27.

After rough adjustments at the common throttles 26 and 27, the amounts of pressure oil to be supplied to the hydrostatic bearings 22A to 22D through the branched oil passages 24B, 25B, 24C and 25C are finely adjusted by the discrete throttles 28 to 31 separately and independently of each other.

Namely, the discrete throttle 28 finely and separately adjusts the amount of pressure oil to be supplied to the first main hydrostatic bearing 22A through the branched oil passage 24B, while the discrete throttle 29 finely and separately adjusts the amount of pressure oil to be supplied to the second main hydrostatic bearing 22B through the branched oil passage 25B. Further, the discrete throttle 30 finely and separately adjusts the amount of pressure oil to be supplied to the first auxiliary hydrostatic bearing 22C through the branched oil passage 24C, while the discrete throttle 31 finely and separately adjusts the amount of pressure oil to be supplied to the second auxiliary hydrostatic bearing 22D through the branched oil passage 25C.

Indicated at 32 and 33 are a pair of tilting actuators which drive the swash plate 21 into a tilted position. In this instance, as shown in Figs. 2, 3, 6 and 7, one tilting actuator 32 is constituted by a cylinder bore 32A which is formed in the main casing 11A and located on a radially outer

side of the cylinder block 14, a tilting piston 32C which is slidably fitted in the cylinder bore 32A, defining a pressure chamber 32B within the cylinder bore 32A, and a spring 32D which is placed in the pressure chamber 32B to constantly bias the tilting piston 32C toward the swash plate 21.

Similarly to the above-mentioned of the tilting actuator 32, the other tilting actuator 33 is constituted by a cylinder bore 33A which is formed in the main casing 11A on a radially outer side of the cylinder block 14, a tilting piston 33C which is slidably fitted in the cylinder bore 33A, defining a pressure chamber 33B within the cylinder bore 33A, and a spring 33D which is placed in the pressure chamber 33B to constantly bias the tilting piston 33C toward the swash plate 21.

In this instance, the tilting actuators 32 and 33 are located in radially opposite portions of the main casing 11A on the outer side of the cylinder block 14 to tilt the swash plate 21 in the direction of arrow A or B by the tilting pistons 32C and 33C. More particularly, as shown in Figs. 3 and 10, the pressure chamber 32B of the tilting actuator 32 is connected to and supplied a tilting control pressure to and from a control conduit 50B, which will be described hereinafter. The pressure chamber 33B of the tilting actuator

33 is connected to and supplied a tilting control pressure to and from a control conduit 50A, which will also be described later on.

When the tilting piston 33C is extended out of the cylinder bore 33A by a tilting control pressure as shown in Fig. 7, the swash plate 21 is tilted by the tilting piston 33C in the direction of arrow A (in a forward direction). At this time, the other tilting piston 32C is retracted into the cylinder bore 32A. When the tilting piston 32C is extended out of the cylinder bore 32A, the swash plate 21 is tilted by the tilting piston 32C in the direction of arrow B (in a reverse direction), and at this time the tilting piston 33C is retracted into the cylinder bore 33A.

Indicated at 34 is a regulator as a volume control valve which supplies a tilting control pressure to and from the tilting actuators 32 and 33. As shown in Fig. 3, this regulator 34 is constituted by a valve housing 35 which is provided on an outer side of the main housing 11A of the casing 11, a control sleeve 36, a spool 37, a hydraulic pilot portion 38 and a valve spring 39, which will be described hereinafter. As shown in Fig. 10, the regulator 34 is constituted by a tilting control hydraulic servo valve having the spool 37 within the control sleeve 36.

In this instance, as shown in Fig. 3, inlet/outlet ports 35A and 35B of the tilting control pressure are provided in the valve housing 35 of the regulator 34. The inlet/outlet port 35A is connected to the discharge side of a pilot pump 46 through a control conduit 48A, which will be described hereinafter. On the other hand, the inlet/outlet port 35B is connected to a control conduit 48B, which will also be described hereinafter. The valve housing 35 of the regulator 34 is provided liquid tight on the outer side of the casing 11. Further, the control sleeve 36 and spool 37 are extended in parallel relation with the rotational shaft 13 (the axis O-O in Fig. 10).

Indicated at 36 is the cylindrical control sleeve which is slidably fitted in the valve housing 35. At one axial end of the control sleeve 36, a translation bar 44 is integrally fixed to the outer periphery of the control sleeve 36 by the use of a plural number of set screws or the like. The control sleeve 36 is put in a sliding displacement in the valve housing 35 in an axial direction (in the direction of arrow D or E in Fig. 6), following a movement of the translation bar 44 (a translational movement in the axial direction of the rotational shaft 13).

Indicated at 37 is a spool which is slidably fitted in

the control sleeve 36. This spool 37 is slidingly displaced in an axial direction of the valve housing 35 along the inner periphery of the control sleeve 36. By displacements of the spool 37, the inlet/outlet port 35B is selectively brought into and out of communication with the inlet/outlet port 35A or the drain passage 11E.

Indicated at 38 is a hydraulic pilot portion which is provided in the valve housing 35 located at one axial end of the spool 37. This hydraulic pilot portion 38 is provided with a plunger 38A for driving the spool 37 in an axial direction against the action of valve spring 39, which will be described hereinafter, and supplied with a command pressure through a command pressure conduit 53 which will also be described later on.

As soon as a command pressure from the command pressure conduit 53 is received at the plunger 38A of the hydraulic pilot portion 38 as a pilot pressure, the spool 37 is put in a sliding displacement in an axial direction within the valve housing 35 according to the received pilot pressure. As a result, by the plunger 38A of the hydraulic portion 38, the regulator 34 is switched to a switched position (II) or (III) from a neutral position (I) of Fig. 10.

Denoted at 39 is a valve spring which is interposed

between the other axial end of the spool 37 and the valve housing 35. This valve spring 39 constantly urges the spool 37 toward the hydraulic pilot portion 38 to return the regulator 34 to a neutral position (I) shown in Fig. 10.

Indicated at 40 is a feedback mechanism to let the regulator 34 follow tilting movements of the swash plate 21 for the purpose of feedback control. As shown in Figs. 3 to 13, the feedback mechanism 40 is constituted by a conversion mechanism 41 and a translation bar 44 which are provided between a lateral side of the swash plate 21 and the control sleeve 36 of the regulator 34.

Indicated at 41 is a conversion mechanism which converts a tilting movement of the swash plate 21 into an axial displacement by means of a cam groove 42 and a cam follower 43 as described below. More particularly, by the conversion mechanism 41, a tilting movement of the swash plate 21 is converted into an axial displacement as described later and generated a translational movement (a straight parallel movement) of a translation bar 44 in the direction of axis O-O of the rotational shaft 13.

Denoted at 42 is a cam groove which is provided with a cam surface for converting a tilting movement of the swash plate 21 into an axial displacement of a cam follower 43. As

shown in Figs. 3 to 8, the cam groove 42 is constituted by a bent groove which is formed substantially in the shape of "V" or "U" on a lateral side of the swash plate 21 (on a lateral side of the other leg portion 21B). The cam groove 42 located at a position which is spaced from the tilting center C of the swash plate 21. Further, in order to receive a roller 43A of the cam follower 43 slidably (rotatably), the cam groove 42 is formed in a width which corresponds to outside diameter of the roller 43A.

In this instance, the cam groove 42 is composed of an intermediate groove portion 42A as a neutral position sliding contact part where the roller 43A of the cam follower 43 is sliding contact when the swash plate 21 is in a zero angle neutral position as shown in Figs. 10 and 11, a downwardly inclined groove portion 42B as a forward direction sliding contact part to be held in sliding contact with the roller 43A of the cam follower 43 when the swash plate 21 is tilted in the direction of arrow A (in a forward direction) from the neutral position, and an upwardly inclined groove portion 42C as a reverse direction sliding contact part to be held in sliding contact with the cam follower roller 43A when the swash plate 21 is tilted in the direction of arrow B (in a reverse direction) form the neutral position.

Of the three groove portions 42A to 42C of the cam groove 42, the intermediate groove portion 42A is located at a most distant position Ra (Ra > R) from the tilting center C of the swash plate 21 along the axis O-O of the rotational shaft 13 when the swash plate 21 is in the neutral position. From the intermediate groove portion 42A, the downwardly inclined groove portion 42B is inclined obliquely downward in a direction toward the tilting center C, and the upwardly inclined groove portion 42C is inclined obliquely upward in a direction toward the tilting center C.

Namely, on a lateral surface of the swash plate 21, the cam groove 42 is formed as a bent groove substantially in the shape of a letter "V" or "U" from the intermediate groove portion 42A, and the downwardly inclined groove portion 42B and the upwardly inclined groove portion 42C are each spread downward and upward from the intermediate groove portion 42A and formed symmetrical figures against the axis O-O.

Fore ends of the downwardly inclined groove portion 42B and the upwardly inclined groove portion 42C are extended as far as points G1 and H1 of Fig. 11, respectively, which are at a distance Rb from the tilting center C of the swash plate 21. In this instance, the distance Rb is smaller than the distance Ra between the tilting center C and the intermediate groove

portion 42A (Rb < Ra < R).

Indicated at 43 is a cam follower which is engaged in the cam groove 42 for sliding contact with the latter. As shown in Fig. 3, this cam follower 43 is integrally provided at one longitudinal end of a translation bar 44, which will be described hereinafter, and provided with a roller 43A which rolls (rotates on its own axis) along a wall surface (cam surface) within the cam groove 42.

Through the roller 43A of the cam follower 43 which is slidably engaged in the cam groove 42 on the side of the swash plate 21, the cam follower 43 converts a tilting movement of the swash plate 21 into an axial displacement and generates a translational movement (a straight parallel movement) of the translation bar 44 along the axis O-O of the rotational shaft 13.

In this instance, when the swash plate 21 is in the neutral position, the roller 43A of the cam follower 43 which is in engagement with the cam groove 42 on the side of the swash plate 21 is located in an initial position of Fig. 11 together with the translation bar 44 on a line F-F perpendicularly intersecting the axis O-O of the rotational shaft 13. At this time, the roller 43A of the cam follower 43 is located at a most receded position along the axis O-O of

the rotational shaft 13 (in the direction of arrow E in Fig. 10).

Further, if the swash plate 21 in the neutral position is tilted in the direction of arrow A (in a forward direction) as shown in Figs. 7 and 12 until its tilting angle θ reaches α ($\theta = \alpha$), the roller 43A of the cam follower 43 is moved in sliding contact with the downwardly inclined groove portion 42B of the cam groove 42 as far as point G1 of Fig. 12. As a consequence, the roller 43A of the cam follower 43 is put in a parallel movement (translational movement) as far as a position on line G-G together with the translation bar 44, making a displacement of a distance a in the axial direction of the rotational shaft 13 from the initial position on line F-F.

On the other hand, if the swash plate 21 in the neutral position is tilted in the direction of arrow B (in a reverse direction) as shown in Fig. 13 until its tilting angle θ reaches β ($\theta = \beta$), the roller 43A of the cam follower 43 is moved in sliding contact with the upwardly inclined groove portion 42C of the cam groove 42 as far as point H1 of Fig. 13. As a result, the roller 43A of the cam follower 43 is put in a parallel movement as far as a position on line H-H together with the translation bar 44 making a displacement of

a distance b in the axial direction of the rotational shaft 13 from the initial position on line F-F.

When the swash plate 21 is tilted through the same tilting angle θ (e.g., α or β) in a forward or reverse direction, the forward and reverse tilting angles α and β are same angles in opposite directions ($\alpha = \beta$), and resulting axial displacements a and b are of the same values (a = b).

Indicated at 44 is a translation bar which serves as a displacement transmission member in the feedback mechanism 40. As shown in Fig. 3, this translation bar 44 is slidably received in the slot 11D of the main casing 11A through a guide member 45, which will be described hereinafter, for translational movements along the axial direction of the rotational shaft 13 (in the direction of axis O-O shown in Fig. 10). As shown in Fig. 3, between a lateral side of the swash plate 21 and the control sleeve 36, one end of the translation bar 44 is extended into the casing 11 radially of the rotational shaft 13 while the other end of the translation bar 44 is extended radially toward the control sleeve 36.

In this instance, the translation bar 44 is provided with the cam follower 43 at one longitudinal end and put in a translational movement together with the cam follower 43 in the direction of the axis O-O of the rotational shaft 13.

Further, as shown in Figs. 3 and 4, the translation bar 44 is provided with a bifurcated anchor portion 44A at the other longitudinal end, which is adapted to grip the control sleeve 36 radially from outside. The anchor portion 44A is securely fixed on the outer periphery of the control sleeve 36 by means of a plural number of set screws, rivets or other suitable fixation means.

More specifically, the translation bar 44 is fixed to the control sleeve 36 at a predetermined angle relative to the latter (e.g., vertically at right angles), and the translation bar 44 is put in an axial displacement along the axis O-O of the rotational shaft 13 along with the roller 43A of the cam follower 43.

In this manner, as the swash plate 21 is tilted in the direction of arrow A or B in Fig. 2, the translation bar 44 of Fig. 3 is put in a parallel movement together with the cam follower 43 in the axial direction of the rotational shaft 13, according to a tilting angle of the swash plate 21. The parallel movement of the translation bar 44 is directly transmitted to the control sleeve 36 of the regulator 34 from the anchor portion 44A to displace the control sleeve 36 in the direction of arrow D or E in Fig. 6 along the axis O-O of the rotational shaft 13. Thus, the translation bar 44 plays

the role of feedback control for the regulator 34.

Indicated at 45 is a guide member which is provided in such a way as to cover the slot 11D of the casing 11 as shown in Fig. 3. In this instance, the guide member 45 is arranged to displacably and slidably support a longitudinally intermediate portion of the translation bar 44, suppressing upward and downward movements (e.g., movements in a radial direction of the cylinder block 14) as well as rattling vibrational movements of the translation bar 44. That is to say, the guide member 45 ensures smooth parallel movements (translational movements) of the translation bar 44 in the axial direction of the rotational shaft 13.

Denoted at 46 is a pilot pump which generates a tilting control pressure. Together with the hydraulic pump 1, this pilot pump 46 is rotationally driven at the prime mover 2 shown in Fig. 1 to deliver a tilting control pressure to a control conduit 48A while sucking in operating oil, for example, from a tank 47 shown in Fig. 3.

In this case, by means of a relief valve 49, the level of pressure oil which is delivered from the pilot pump 46 is maintained at a sufficiently lower level as compared with the output pressure of the hydraulic pump 1. A control conduit 48B is provided between the inlet/outlet port 35B of the

regulator 34 and a forward/reverse switch valve 51 which will be described hereinafter.

Indicated at 50A and 50B are other control conduits which supply a tilting control pressure to and from the pressure chambers 32B and 33B of the tilting actuators 32 and 33, respectively. As shown in Figs. 3 and 10, by switching operations of a forward/reverse switch valve 51, the control conduits 50A and 50B are brought into and out of communication with the control conduits 48A and 48B, respectively.

Indicated at 51 is a forward/reverse switch valve as a directional control valve which is provided between the control conduits 48A and 48B and the secondary control conduits 50A and 50B. As shown in Figs. 3 and 10, this forward/reverse switch valve 51 is provided with right and left solenoids 51A and 51B and can be switched to and a forward position (b) or a reverse position (c) from a vehicle stop position (a) or vice versa by, for example, manually operating a switch lever (not shown) which is provided in a cab or the like.

When the forward/reverse switch valve 51 is switched to the forward position (b) from the stop position (a), a tilting control pressure is supplied to the pressure chamber 33B of the tilting actuator 33 from the pilot pump 46 through the

control conduits 48A and 50A, according to the degree of depression of a vehicle drive pedal 52A by an operator, which will be described later.

At this time, a tilting control pressure in the pressure chamber 32B of the tilting actuator 32 is discharged to the tank 47 through the control conduits 50B and 48B and the regulator 34, and the swash plate 21 is driven by the tilting piston 33C of the tilting actuator 33 to tilt in the direction of arrow A in Fig. 10.

On the other hand, when the forward/reverse switch valve 51 is switched to the reverse position (c) from the stop position (a), a tilting control pressure is supplied to the pressure chamber 32B of the tilting actuator 32 from the pilot pump 46 through the control conduits 48A and 50B, according to the degree of depression of the vehicle drive pedal 52A by an operator. At the same time, a tilting control pressure in the pressure chamber 33B of the tilting actuator 33 is discharged to the tank 47 through the control conduits 50A and 48B and the regulator 34. As a consequence, the swash plate 21 is driven by the tilting piston 32C of the tilting actuator 32 to tilt in the direction of arrow B in Fig. 10.

In this manner, the forward/reverse switch valve 51 is provided between the regulator 34 and the tilting actuators 32

and 33, and it is switched from a stop position (a) to a forward position (b) or a reverse position (c) to change over the direction of tilting control pressure to the tilting actuator 32 or to the tilting actuator 33 thereby to drive the swash plate 21 to tilt in a forward or reverse direction from a neutral position according to a tilting control pressure.

Indicated at 52 is a vehicle control valve which is provided in the side of a driver room of a wheel type vehicle as a command means. Attached to the vehicle control valve 52 is a vehicle drive pedal 52A which corresponds to an accelerator pedal of an automotive vehicle. As the vehicle drive pedal 52A is depressed by an operator of the vehicle, a pilot pressure is supplied to the hydraulic pilot portion 38 of the regulator 34 as a command signal through a command pressure conduit 53, variably adjusting the vehicle speed of the automotive vehicle in the manner as described hereinafter.

Being constructed as described above, the swash plate type variable displacement hydraulic pump 1 according to the present embodiment operates in a vehicle driving hydraulic circuit of a wheel type working vehicle, in the manner as follows.

In the first place, when the forward/reverse switch valve 51 is in the stop position (a) shown in Fig. 10, both of the

control conduits 50A and 50B are connected to the control conduits 48A, the pressure chambers 32B and 33B of the tilting actuators 32 and 33 are maintained at the same pressure level to retain the swash plate 21 in the zero angle neutral position.

Therefore, even if the cylinder block 14 is put in rotation by rotationally driving the rotational shaft 13 by the prime mover 2, the pistons 16 are not reciprocated within the respective cylinders 15 of the cylinder block 14, and the supply/discharge passages 12A and 12B of the hydraulic pump 1 remain at the same pressure, stopping supply of pressure oil to and from the hydraulic motor 5 in Fig. 1 through the main conduits 3A and 3B.

In the next place, if the forward/reverse switch valve 51 is switched to the forward position (b) from the stop position (a) and the vehicle drive pedal 52A is depressed by an operator, pressure oil from the pilot pump 46 is supplied to the pressure chamber 33B of the tilting actuator 33 through the control conduits 48A and 50A.

At this time, according to the degree of depression of the vehicle drive pedal 52A, a pilot pressure is supplied toward the hydraulic pilot portion 38 of the regulator 34 from the command pressure conduit 53. Whereupon, within the valve

housing 35 of the regulator 34, the spool 37 is put in an axial sliding displacement according to the pilot pressure, switching the regulator 34 from the neutral position (I) to the switch position (II) shown in Fig. 10.

Therefore, the control conduit 48B is now connected to the tank 47 through the regulator 34 and the drain chamber within the casing 11 to discharge pressure oil in the pressure chamber 32B of the tilting actuator 32 to the tank 47 through the control conduits 50B and 48B and the regulator 34. As a result, the swash plate 21 is driven by the piston 33C of the tilting actuator 33 in the direction of arrow A in Fig. 10.

When the swash plate 21 is tilted in the direction of arrow A, the cylinder block 14 is put in rotation together with the rotational shaft 13, the pistons 16 are repeatedly reciprocated within the respective cylinders 15 of the cylinder block 14, with a stroke length (displacement volume) commensurate with the tilting angle θ of the swash plate 21. At this time, for example, the hydraulic pump 1 takes oil into the cylinders 15 from the side of the supply/discharge passage 12B while discharging pressure oil from the supply/discharge passage 12A.

Thus, in the vehicle driving closed hydraulic circuit 4 of Fig. 1, pressure oil is circulated within the main conduits

3A and 3B along the direction of arrow A1, and rotationally driving the vehicle driving hydraulic motor 5 by pumping pressure oil thereto. The rotational output of the hydraulic motor 5 is transmitted to the wheels 7 of the wheel type working vehicle through the reducer 6 and the wheels 7 are rotationally driven, for example, to drive the working vehicle in a forward direction at a speed commensurate with the tilting angle θ of the swash plate 21.

On the other hand, if the forward/reverse switch valve 51 in the stop position (a) is switched to the reverse position (c) and the vehicle drive pedal 52A is depressed by an operator, the regulator 34 which is in the neutral position (I) of Fig. 10 is switched to the switched position (II). Whereupon, pressure oil from the pilot pump 46 is supplied to the pressure chamber 32B of the tilting actuator 32 through the control conduits 48A and 50B. At the same time, pressure oil in the pressure chamber 33B of the tilting actuator 33 is discharged to the tank 47 through the control conduits 50A and 48B and the regulator 34. Thus, the swash plate 21 is driven by the piston 32C of the tilting actuator 32 to tilt in the direction of arrow B in Fig. 10.

Further, in this case, pressure oil is circulated in the vehicle driving closed hydraulic circuit 4 in Fig. 1 through

the direction of arrow B1, rotationally driving the vehicle driving hydraulic motor 5 in the same direction. Consequently, the rotational output of the hydraulic motor 5 is transmitted to wheels 7 of the wheel type working vehicle through the reducer 6, for example, to drive the working vehicle in a reverse direction at a speed commensurate with the tilting angle θ.

In this instance, when the swash plate 21 is tilted in the forward direction (in the direction A) from the neutral position, one of a pair of supply/discharge passages 12A and 12B, the supply/discharge passage 12A turns to a high pressure, and the swash plate 21 receives the resultant force f1 of hydraulic reaction forces of the respective pistons 16 at the position of the resultant force acting point k1 shown in Fig. 5.

However, from the supply/discharge passage 12A and through the common oil passage 24A, branched oil passages 24B and 24C of the oil guide passage 24, high pressure oil is drawn to the first main hydrostatic bearing 22A which is provided on the leg portion 21A of the swash plate 21, and to the first auxiliary hydrostatic bearing 22C on the leg portion 21B as well. Therefore, between the tilting support surfaces 20A and 20B of the swash plate support member 20 and the leg

portions 21A and 21B of the swash plate 21, a dissociative force fa is generated by the first main hydrostatic bearing 22A and a dissociative force fc is generated by the first auxiliary hydrostatic bearing 22C.

As shown in Fig. 5, the first main hydrostatic bearing 22A is located at a distance La from the resultant force acting point k1 of hydraulic reaction forces exerted on the swash plate 21 from the respective pistons 16, and the first auxiliary hydrostatic bearing 22C is located at a distance Lc (Lc > La) from the resultant force acting point k1.

Thus, with regard to the resultant force f1 of the hydraulic reaction forces which is exerted on the swash plate 21 from the respective pistons 16, the dissociative force fa of the first main hydrostatic bearing 22A and the dissociative force fc of the first auxiliary hydrostatic bearing 22C should be set in such a way as to satisfy the relations expressed by Formulas (1) to (4) below. If arranged in this way, the resultant force f1 of hydraulic reaction forces can be balanced with the dissociative forces fa and fc, so that the contacting surface of the tilting support surfaces 20A and 20B of the swash plate support member 20 and the leg portions 21A and 21B of the swash plate 21 can be maintained in a lubricated state.

Namely, with regard to the resultant force f1 of hydraulic reaction forces which are exerted on the swash plate 21 tilted from neutral position in forward direction by the pistons 16, the dissociative force fa of the first main hydrostatic bearing 22A and the dissociative force fc of the first auxiliary hydrostatic bearing 22C are set to satisfy conditions of the following Formula.

$$f1 \fallingdotseq fa + fc \quad \ldots\ldots (1)$$

On the basis of pressure P of hydraulic reaction forces and pressure receiving surface area S1, the resultant force f1 which is exerted on the swash plate 21 can be expressed by the following Formula.

$$f1 = S1 \times P \quad \ldots\ldots (2)$$

Assuming a case that a similar pressure P is exerted on the first main hydrostatic bearing 22A and the first auxiliary hydrostatic bearing 22C, bearings 22A and 22B have effective bearing surface areas Sa and Sc (Sc < Sa), respectively, so that the relation of the following Formula can be led from the Formulas (1) and (2).

$$S1 \fallingdotseq Sa + Sc \quad \ldots\ldots (3)$$

The dissociative force fa of the first main hydrostatic bearing 22A (an effective bearing surface area Sa) acts at a position which is at a distance La from the resulting force

acting point k1, while the dissociative force fc of the first auxiliary hydrostatic bearing 22C (an effective bearing surface area Sc) acts at a position which is at a distance Lc from the resultant force acting point k1. Therefore, on the basis of the resultant force acting point k1, moments of the dissociative forces fa and fc are set to satisfy the relations of the following Formula.

$$La \times Sa \fallingdotseq Lc \times Sc \quad \text{......................(4)}$$

By the foregoing settings, relative to the resultant force f1 which is exerted on the swash plate 21 by hydraulic reaction forces of the pistons 16, the dissociative force fa of the main hydrostatic bearing 22A and the dissociative force fc of the auxiliary hydrostatic bearing 22C are balanced, preventing the leg portions 21A and 21B of the swash plate 21 from floating up away from the tilting support surfaces 20A and 20B of the swash plate support member 20.

As a consequence, the pressure oil which is drawn to the hydrostatic bearings 22A and 22C are prevented from leaking to the outside, that is to say, the leg portions 21A and 21B of the swash plate 21 as well as the tilting support surfaces 20A and 20B of the swash plate support member 20 are maintained in a lubricated state, stabilizing tilting motions of the swash plate 21 and permitting to minimize the rotational driving

force of the tilting actuators 32 and 33.

On the other hand, when the swash plate 21 is tilted in a reverse direction (direction B) from the neutral position, a resultant force f2 of hydraulic reactions forces of the pistons 16 is exerted on the swash plate 21 at a position of resultant force acting point k2 shown in Fig. 5. With regard to this resultant force f2, the dissociative force fb of the second main hydrostatic bearing 22B and the dissociative force fd of the second auxiliary hydrostatic bearing 22D are set to satisfy relations of the following Formula.

$$f2 \fallingdotseq fb + fd \quad \ldots\ldots (5)$$

On the relation of pressure P of hydraulic reaction forces and pressure receiving surface area S2, the resultant force f2 which is exerted on the swash plate 21 at this time can be expressed as in the following Formula.

$$f2 = S2 \times P \quad \ldots\ldots (6)$$

Assuming a case that a similar pressure P acts on the second main hydrostatic bearing 22B and the second auxiliary hydrostatic bearing 22D, bearings 22B and 22D have effective bearing surface areas Sb and Sd ($Sd < Sb$), respectively, so that the relations of the following Formula are led from the Formula (5).

$$S2 \fallingdotseq Sb + Sd \quad \ldots\ldots (7)$$

The dissociative force fb of the second main hydrostatic bearing 22B (an effective bearing surface area Sb) acts at a position which is at a distance of Lb from the resultant force acting point k2, while the dissociative force fd of the second auxiliary hydrostatic baring 22D (an effective bearing surface area Sd) acts at a position which is at a distance of Ld from the resultant force acting point k2. Therefore, on the basis of the resultant force acting point k2, moments of the dissociative forces fb and fd are set to satisfy relations of the following Formula.

$$Lb \times Sb \fallingdotseq Ld \times Sd \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(8)$$

With the foregoing settings, relative to the resultant force f2 which is exerted on the swash plate 21 by hydraulic reaction forces of the pistons 16, the dissociative force fb of the main hydrostatic bearing 22B and the dissociative force fd of the auxiliary hydrostatic bearing 22D are balanced, preventing the leg portions 21A and 21B of the swash plate 21 from floating up away from the tilting support surfaces 20A and 20B of the swash plate support member 20.

As a consequence, even when the swash plate 21 is tilted in a reverse direction from the neutral position, pressure oil which has been drawn into the hydrostatic bearings 22B and 22D is prevented from leaking to the outside, that is to say, the

leg portions 21A and 21B of the swash plate 21 as well as the tilting support surfaces 20A and 20B of the swash plate support member 20 are maintained in a lubricated state, stabilizing tilting motions of the swash plate 21 and permitting to minimize the rotationally driving force of the tilting actuators 32 and 33.

By the way, the traveling speed of the vehicle in a forward or reverse direction is determined by a pressure oil discharge rate (flow rate) of the hydraulic pump 1, which is increased or decreased according to the tilting angle θ of the swash plate 21. The regulator 34 is a volume control valve, so that, unless operated under feedback control corresponding to the tilting angle θ, it is difficult to control the tilting angle θ of the swash plate 21 (i.e., the traveling speed of the vehicle) in stable conditions simply by depressing the vehicle drive pedal 52A.

Therefore, in the present embodiment, a feedback mechanism 40 is provided between the control sleeve 36 of the regulator 34 and a lateral side of the swash plate 21. For feedback control of the regulator 34, this feedback mechanism 40 is arranged to make the regulator 34 follow tilting movements of the swash plate 21 when the swash plate 21 is driven to tilt either in a forward direction or in a reverse

direction from the zero angle neutral position.

The feedback mechanism 40 is constituted by the cam groove 42 which is formed as a groove in "V" or "U" bent shape on a lateral side of the swash plate 21 (on a lateral side of the leg portion 21B) on the basis of the axis O-O of the rotational shaft 13, the cam follower 43 having the roller 43A in sliding contact with the cam groove 42 to pick up a tilting motion of the swash plate 21 as an axial displacement, and the translation bar 44 which is moved parallel to the axial direction of the rotational shaft 13 by the axial displacement of the cam follower 43. Thus, by the anchor portion 44A at the distal end of the translation bar 44, the axial displacement of the cam follower 43 is transmitted to the control sleeve 36.

In this case, the cam groove 42 on the side of the swash plate 21 is composed of the intermediate groove portion 42A which is located at a most distant position Ra ($Ra < R$) from the tilting center C of the swash plate 21 along the axis O-O of the rotational shaft 13 when the swash plate 21 is in the neutral position as shown in Fig. 11, the downwardly inclined groove portion 42B which is inclined obliquely downward from the intermediate groove portion 42A in a direction toward the tilting center C, and the upwardly inclined groove portion 42C

which is inclined obliquely upward from the intermediate groove portion 42A in a direction toward the tilting center C. As a whole, the cam groove 42 is formed as a groove which is bent in the shape of letter "V" of "U" at the position of the intermediate groove portion 42A of the lateral side of the swash plate 21.

Further, since anchor portion 44A of the translation bar 44 is vertically fixed to the control sleeve 36, the roller 43A of the cam follower 43 is restricted of movements (deviational movements) in a direction perpendicular to the axis O-O of the rotational shaft 13 and permitted of only an axial displacement along the axis O-O.

When the swash plate 21 is located in the zero angle neutral position, the roller 43A of the cam follower 43 is located in the intermediate groove portion 42A of the cam groove 42. When the swash plate 21 is tilted in the direction of arrow A (in a forward direction) from the neutral position, the roller 43A is moved along and in sliding contact with the downwardly inclined groove portion 42B. Conversely, when the swash plate 21 is tilted in the direction of arrow B (in a reverse direction) from the neutral position, the roller 43A is moved along and in sliding contact with the upwardly inclined groove portion 42C.

Therefore, when the swash plate 21 is tilted in the direction of arrow A (in a forward direction) from the neutral position as shown in Fig. 12 until the tilting angle θ reaches α ($\theta = \alpha$), the roller 43A of the cam follower 43 is slid along the downwardly inclined groove portion 42B of the cam groove 42 as far as a position at point G1, and at the same time the translation bar 44 is put in a parallel movement (a translational movement) together with the cam follower 43 as far as a position on line G-G of Fig. 12.

Line G-G which passes the point G1 is located at a distance Rb from the tilting center C. On the other hand, when the swash plate 21 is in the neutral position, the translation bar 44 is located in an initial position on line F-F which is at a distance Ra from the tilting center C of the swash plate 21. Thus, the distance of axial displacement of the translation bar 44 from the initial position on line F-F to the position on line G-G in the axial direction of the rotational shaft 13 can be obtained as a distance $\underline{a}$ by the following Formula (9).

$$\underline{a} = Ra - Rb \quad \cdots\cdots (9)$$

On the other hand, when the swash plate 21 is tilted in the direction of arrow B (in a reverse direction) from the neutral position until its tilting angle θ reaches β ($\theta = \beta$)

as shown in Fig. 13, the roller 43A of the cam follower 43 is slid along the upwardly inclined groove portion 42C of the cam groove 42 as far as a position H1, and at the same time the translation bar 44 is put in a parallel movement together with the cam follower 43 as far as a position on line H-H of Fig. 13.

In this case, line H-H which passes the point H1 is also at a distance Rb from the tilting center C. Thus, the distance of axial displacement of the translation bar 44 from the initial position on line F-F to the position on line H-H in the axial direction of the rotational shaft 13 can be obtained as a distance $\underline{b}$ by the following Formula (10).

$$\underline{b} = Ra - Rb \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (10)$$

In this manner, by the cam follower 43 which has the roller 43A in sliding contact with the cam groove 42 on the side of the swash plate 21, a tilting movement of the swash plate 21 in a forward or reverse direction is converted into an axial displacement of the translation bar 44 along the direction of axis O-O of the rotational shaft 13 (e.g., a displacement over a distance $\underline{a}$ or $\underline{b}$), and this axial displacement of the translation bar 44 is directly transmitted to the control sleeve 36 as same axial displacement through the anchor portion 44A at the distal end of the translation

bar 44.

Thus, according to the present embodiment, even in a case where the swash plate type variable displacement hydraulic pump 1 is connected to a hydraulic motor 5 through a closed hydraulic circuit 4 as shown in Fig. 1, the discharge rate (flow rate) of pressure oil can be controlled in both forward and reverse directions by tilting the swash plate 21, a variable displacement portion, in a forward or reverse direction from a zero angle neutral position when automobile is driven in a forward direction and backward direction, and speed of the automobile can be controlled smoothly corresponding to the tilting angle of the swash plate 21.

In addition, the regulator 34 which functions as a volume control valve can be constituted by a simple hydraulic servo valve having the spool 37 within the control sleeve 36. This means that the tilting control system, which is constituted by the tilting actuators 32 and 33, regulator 34 and feedback mechanism 40, can be simplified in construction as a whole and can be assembled efficiently by the use of a reduced number of parts.

Further, providing the forward/reverse switch valve 51 between the regulator 34 and the tilting actuators 32 and 33, the tilting control system including the regulator 34 can be

simplified in construction as a whole as compared with a prior art counterpart and can be manufactured with a higher productivity and at a lower cost.

Moreover, the tilting control system of the hydraulic pump 1 can be applied not only to the closed hydraulic circuit 4 as exemplified in Fig. 1, but also to the so-called open hydraulic circuit for supplying pressure oil to and from a hydraulic actuator like a hydraulic motor. That is to say, the tilting control system of the hydraulic pump 1 is a versatile system which can be applied to both closed and open hydraulic circuits, and can be manufactured with a higher productivity and at a lower cost.

Further, according to the present embodiment, hydrostatic bearings 22 (hydrostatic bearings 22A to 22D) are provided between the tilting support surfaces 20A and 20B of the swash plate support member 20 and the leg portions 21A and 21B of the swash plate 21, drawing high pressure oil to the respective hydrostatic bearings 22A to 22D from a pair of supply/discharge passages 12A and 12B. By the hydrostatic bearings 22A to 22D, dissociative forces (e.g., dissociative forces fa, fb, fc and fd in Fig. 5) are generated between the tilting support surfaces 20A and 20B and the leg portions 21A and 21B to maintain the contacting surfaces in a lubricated

state.

As a consequence, the dissociative forces fa and fc of the main hydrostatic bearings 22A and 22B (the dissociative forces fb and fd of the auxiliary hydrostatic bearings 22C and 22D) are kept in a well-balanced state relative to the resultant force f1 (the resultant force f2) of hydraulic reaction forces which are exerted on the swash plate 21 by the pistons 16, so that the hydrostatic bearings 22 constituted by the hydrostatic bearings 22A to 22D can be stabilized in performance as a hydrostatic bearing.

Thus, in addition to the swash plate type variable displacement hydraulic pump 1 for use in HST, the present invention can be easily applied, for example, to a hydraulic motor with a reversible rotational shaft or to hydraulic rotary machines having a pair of supply/discharge passages which are reversibly switched to high-pressure and low-pressure sides, for manufacturing a hydraulic rotary machine of higher versatility at a reduced cost and with higher productivity.

As shown in Fig. 5, the first and second main hydrostatic bearings 22A and 22B are located near the acting points k1 and k2 of the resultant force of hydraulic reaction forces which are exerted on the swash plate 21 by the pistons 16. That is

to say, the dissociative forces fa and fb of the main hydrostatic bearings 22A and 22B can be close to the resultant force acting points k1 and k2.

The closeness of the acting points of dissociative forces fa and fb to the resultant force acting points k1 and k2 makes it possible to diminish a moments which acts on the swash plate 21 (e.g., rotary moment induced at the resultant force acting points k1 and k2). As a result, it becomes possible to reduce effective bearing surface areas Sc and Sd of the first and second auxiliary hydrostatic bearings 22C and 22D and to downsize the hydraulic pump 1 into a compact form as a whole including the swash plate 21.

Further, first and second slide bearings 23A and 23B are provided on the leg portions 21A and 21B of the swash plate 21, at the positions where are more distant from the rotational shaft 13 than the auxiliary hydrostatic bearings 22D and 22C in the radial direction. Therefore, stability of the swash plate 21 is guaranteed by these slide bearings 23A and 23B even when changes occur to the balance of moments which act on the swash plate 21 by the change of the pressure on the side of the supply/discharge passages 12A and 12B.

Besides, the first and second slide bearings 23A and 23B are held in sliding contact with the tilting support surfaces

20A and 20B of the swash plate support member 20 under a small surface pressure. This means that the slide bearings 23A and 23B makes it possible to reduce surface pressures between the leg portions 21A and 21B of the swash plate 21 and the tilting support surfaces 20A and 20B of the swash plate support member 20, contributing to suppress abrasion of contacting surfaces thereof and to ensure higher reliability and longer service life.

On the other hand, the common oil passage 24A and branched oil passages 24B and 24C are provided between one supply/discharge passage 12A and the first main hydrostatic bearing 22A and the first auxiliary hydrostatic bearing 22C. Besides, another common oil passage 25A and branched oil passages 25B and 25C are provided between the other supply/discharge passage 12B and the second main hydrostatic bearing 22B and the second auxiliary hydrostatic bearing 22D. Moreover, the common throttles 26 and 27 are provided in the course of the common oil passages 24A and 25A, respectively.

Therefore, even if the throttle diameter (orifice diameter) of the common throttles 26 and 27 are formed relatively large diameter, the amounts of pressure oil to be supplied to the main hydrostatic bearings 22A and 22B and the auxiliary hydrostatic bearings 22C and 22D are suitably

adjusted through the common throttles 26 and 27, while reducing possibilities of the common throttles 26 and 27 being clogged (blocked) with dust or other foreign matter to ensure higher reliability of operation.

Further, even if fine gap spaces exist around the hydrostatic bearings 22A to 22D, the common throttles 26 and 27 have effects of preventing oil leaks through such small gaps, improving machining process and productivity of the machine while reducing the manufacturing cost.

In addition, independently discrete throttles 28 to 31 are provided in the branched oil passages 24B, 24C, 25B and 25C. Therefore, by way of these discrete throttles 28 to 31, the amounts of pressure oil to be supplied to the main hydrostatic bearings 22A and 22B and the auxiliary hydrostatic bearings 22C and 22D can be adjusted independently of each other. That is to say, the dissociative forces fa, fb, fc and fd of the hydrostatic bearings 22A to 22D on the swash plate 21 can be easily increased or reduced by way of the flow rate of pressure oil through the respective discrete throttles 28 to 31.

Thus, the balance of moments acting on the swash plate 21 can be enhanced according to the resultant forces f1 and f2 of hydraulic reaction forces, which are exerted on the swash

plate 21 by the pistons 16, and the dissociative forces fa, fb, fc and fd of the hydrostatic bearings 22A, 22B, 22C and 22D, to improve tilting motions and stability of the swash plate 21 while ensuring higher reliability and a prolonged service life as a hydraulic pump 1.

Shown in Figs. 14 to 16 is a second embodiment of the present invention. This embodiment has features in that main and auxiliary hydrostatic bearings on leg portions of a swash plate are spaced apart from each other in a surcumferential direction along a convexly curved surface of the leg portion, and oil passages are bored internally of the swash plate to draw pressure oil into the auxiliary hydrostatic bearings. In the following description of the second embodiment, those component parts which are identical with the counterparts in the first embodiment are simply designated by the same reference numerals or characters to avoid repetitions of the same explanations.

In the drawings, indicated at 61 is a swash plate type variable displacement hydraulic pump adopted in the second embodiment. Substantially in the same manner as the hydraulic pump 1 described in the foregoing first embodiment, the hydraulic pump 61 is largely constituted by the casing 11, the rotational shaft 13, the cylinder block 14, a plural number of

cylinders 15, the pistons 16, the shoes 17, the valve plate 19, the swash plate support member 20 and the swash plate 21.

Indicated at 62 are hydrostatic bearings employed in the present embodiment between the tilting support surfaces 20A and 20B of the swash plate support member 20 and leg portions 21A and 21B of the swash plate 21. In the same manner as the hydrostatic bearings 22 in the first embodiment, pressure oil is drawn into the hydrostatic bearings 62 from a pair of supply/discharge passages 12A and 12B to generate a dissociative force (a hydraulic force) between the tilting support surfaces 20A and 20B and the leg portions 21A and 21B and to keep the contacting surfaces in a lubricated state.

However, in this case, as shown in Figs. 15 and 16, the hydrostatic bearings 62 are composed of a first main hydrostatic bearing 62A which is located on a convexly curved surface of one leg portion 21A at a position close to the through hole 21D of the swash plate 21, a second main hydrostatic bearing 62B which is located on a convexly curved surface of the other leg portion 21B at a position close to the through hole 21D of the swash plate 21, a couple of first auxiliary hydrostatic bearings 62C which are located on the convexly curved surface of the leg portion 21B at circumferentially spaced positions relative to the second main

hydrostatic bearing 62B, and a couple of second auxiliary hydrostatic bearings 62D which are located on the convexly curved surface of the leg portion 21A at circumferentially spaced positions from the first main hydrostatic bearing 62A.

As shown in Fig. 15, the first and second main hydrostatic bearings 62A and 62B are in the form of grooves which are formed on and along convexly curved surfaces of the leg portions 21A and 21B to extend in the directions of arrows A and B and to present an elongated rectangular shape in plane view as shown in Fig. 16. Further, the first auxiliary hydrostatic bearings 62C are in the form of narrow elliptical grooves which are formed and extended widthwise on the convexly curved surface of the leg portion 21B circumferentially on the opposite sides of the second main hydrostatic bearing 62B on and along convexly curved surfaces of the leg portions 21B.

The second auxiliary hydrostatic bearings 62D are in the form of narrow rectangular grooves which are formed widthwise on the convexly curved surface of the leg portion 21A radially on the opposite sides of the first main hydrostatic bearing 62A on and along convexly curved surface of the leg portions 21A.

Of the hydrostatic bearings 62A to 62D, the first main

hydrostatic bearing 62A and the first auxiliary hydrostatic bearings 62C are connected to one supply/discharge passage 12A through an oil guide passage 64 which will be described hereinafter, while the second main hydrostatic bearing 62B and the second auxiliary hydrostatic bearings 62D are connected to the other supply/discharge passage 12B through an oil guide passage 65 which will also be described hereinafter.

The first main hydrostatic bearing 62A is located near a resultant force acting point k1 where a resultant force of hydraulic reaction forces of the pistons 16 is exerted on the swash plate 21 radially on one side of the through hole 21D of the swash plate 21 (on the right side in Fig. 16). On the other hand, the second main hydrostatic bearing 62B is located in the vicinity of a resultant force acting point k2 where a resultant force of hydraulic reactions forces of the pistons 16 is exerted on the swash plate 21 radially on the other side of the through hole 21D of the swash plate 21 (on the left side in Fig. 16).

The main hydrostatic bearings 62A and 62B and the auxiliary hydrostatic bearings 62C and 62D of the present embodiment have substantially the same effective surface areas as the main hydrostatic bearings 22A and 22B and the auxiliary hydrostatic bearings 22C and 22D in the foregoing first

embodiment.

Indicated at 63A and 63B are first and second slide bearings which are provided on the leg portions 21A and 21B of the swash plate 21. These first and second slide bearings 63A and 63B are formed substantially in the same manner as the slide bearings 23A and 23B in the foregoing first embodiment.

Indicated at 64 is an oil guide passage for drawing pressure oil to the hydrostatic bearings 62A and 62C of the hydrostatic bearing 62, and at 65 is the other oil guide passage for drawing pressure oil to the hydrostatic bearings 62B and 62D of the hydrostatic bearing 62. As shown in Figs. 14 to 16, by way of these oil guide passages 64 and 65, the hydrostatic bearings 62A to 62D are connected to a pair of supply/discharge passages 12A and 12B. The one oil guide passage 64 is provided between one supply/discharge passage 12A and the first main hydrostatic bearing 62A and auxiliary hydrostatic bearing 62C, while the other oil guide passage 65 is provided between the other supply/discharge passage 12B and the second main hydrostatic bearing 62B and second auxiliary hydrostatic bearing 62D.

In this instance, one oil guide passage 64 is composed of a first oil passage 64A (see Fig. 14) having one end communicated with the supply/discharge passage 12A and the

other end extended toward the first main hydrostatic bearing 62A, and a second oil passage 64B, a third oil passage 64C and fourth oil passages 64D, which are bored in the swash plate 21. By way of the second oil passage 64B, third oil passage 64C and fourth oil passages 64D, the first main hydrostatic bearing 62A is communicated with the first auxiliary hydrostatic bearings 62C.

In this case, as shown in Figs. 15 and 16, one end of the second oil passage 64B is opened into the first main hydrostatic bearing 62A while the other end is communicated with one end of the fourth oil passages 64D through the third oil passage 64C. The fourth oil passages 64D are branched off in V-shape and opened into the first auxiliary hydrostatic bearings 62C at the respective distal ends.

As shown in Figs. 14 to 16, the other oil guide passage 65 is composed of a first oil passage 65A having one end communicated with the other supply/discharge passage 12B and the other end extended as far as the second main hydrostatic bearing 62B, and a second oil passage 65B, a third oil passage 65C and fourth oil passages 65D, which are bored into the swash plate 21. By way of the second oil passage 65B, third oil passage 65C and fourth oil passage 65D, the second main hydrostatic bearing 62B is communicated with the auxiliary

hydrostatic bearings 62D.

In this case, as shown in Figs. 15 and 16, one end of the second oil passage 65B is opened into the second main hydrostatic bearing 62B while the other end is connected to one end of the fourth oil passages 65D through the third oil passage 65C. The fourth oil passages 65D are branched off in V-shape and opened into the second auxiliary hydrostatic bearings 62D at the respective distal ends.

Indicated at 66 is a throttle which is provided in the course of the first oil passage 64A, and at 67 is an another throttle which is provided in the course of the first oil passage 65A. Of the two throttles 66 and 67, one throttle 66 adjusts the amount of pressure oil to be supplied from the supply/discharge passage 12A to the first main hydrostatic bearing 62A according to its throttle diameter (orifice diameter) as shown in Fig. 14. The other throttle 67 adjusts the amount of pressure oil to be supplied from the supply/discharge passage 12B to the second main hydrostatic bearing 62B according to its throttle diameter (orifice diameter).

In this case, one throttle 66 commonly adjusts the amount of pressure oil to be supplied to the first main hydrostatic bearing 62A and the first auxiliary hydrostatic bearings 62C.

Similarly, the other throttle 67 commonly adjust the amount of pressure oil to be supplied to the second main hydrostatic bearing 62B and the second auxiliary hydrostatic bearings 62D.

Being arranged in the manner as described above, the present embodiment can produce substantially same operational effects as the foregoing first embodiment in stabilizing the tilting actions of the swash plate 21.

According to this particular embodiment, however, the oil passages 64A to 64D are formed internally of the swash plate 21 along with the oil passages 65A to 65D. Therefore, the oil passages 64A and 65A in the casing and the swash plate support member 20 can be simplified in construction to improve working efficiency in manufacturing and machining processes.

In the first embodiment of the invention, by way of example the main hydrostatic bearings 22A and 22B and the auxiliary hydrostatic bearings 22C and 22D are provided on the leg portions 21A and 21B of the swash plate 21. However, the present invention is not limited to the particular example shown. For example, the first and second main hydrostatic bearings and the first and second auxiliary hydrostatic bearings may be provided on the tilting support surfaces 20A and 20B of the swash plate support member 20.

Alternatively, arrangements may be made to provide the

first and second main hydrostatic bearings and the first and second auxiliary hydrostatic bearings on the both tilting support surfaces 20A and 20B of the swash plate support member 20 and the leg portions 21A and 21B of the swash plate 21. In this regard, the same applies to the second embodiment.

Further, in the foregoing first embodiment, for feedback control to let the regulator 34 follow tilting movements of the swash plate 21, the conversion mechanism 41 of the feedback mechanism 40 is constituted by the cam groove 42 and the cam follower 43. However, needless to say, the present invention is not limited to the particular example shown. The conversion mechanism of the feedback mechanism can be made by using a mechanism other than a cam.

Furthermore, in the foregoing embodiments, by way of example the drive control valve 52 is employed as an external command means for supplying a pilot pressure to the regulator 34 as a command signal proportional to the degree of depression of the vehicle drive pedal 52A. However, the present invention is not limited to this particular example. For instance, an electromagnetic proportional solenoid may be employed for the hydraulic pilot portion 38 of the regulator 34, together with an external command means which output an electric signal as a command signal proportional to the degree

of depression of the vehicle drive pedal 52A.

Further, in the foregoing embodiments, by way of example the swash plate type variable displacement hydraulic pump 1 or 61 is applied to a vehicle drive hydraulic circuit of a wheel type working vehicle like a wheel loader. However, it is to be understood that the present invention can be applied not only to a vehicle drive hydraulic circuit but also to other closed hydraulic circuits of various purposes, for example, to a swinging hydraulic circuit.

Furthermore, in the foregoing embodiments, by way of example the swash plate type variable displacement hydraulic rotary machine is applied as a hydraulic pump 1 or 61. However, it is to be understood that application of the present invention is not limited to a swash plate type variable displacement hydraulic pump. For instance, the present invention can be applied to a swash plate type variable displacement hydraulic motor or the like.

Moreover, application of the present invention is not limited to a working vehicle like a wheel loader. For instance, the present invention can be applied to other working vehicles such as wheel type hydraulic excavators, wheel type hydraulic cranes, bulldozers and lift trucks, or to crawler type hydraulic excavators as well.